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(54) Ultra-low fin heat exchanger with enhanced heat transfer.

(57) The present invention is directed to apparatus and a process for heat exchanging wherein ultra-low fins are affixed to the outside of a tubular conduit to enhance heat transfer. The invention has particular application to base load natural gas liquefaction coil-wound heat exchangers.

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ULTRA-LOW FIN HEAT EXCHANGER WITH ENHANCED HEAT TRANSFER

TECHNICAL FIELD

The present invention is directed to an apparatus and process for the enhancement of heat transfer in a heat exchanger. More specifically, the present invention is directed to the use of ultra-low fins on conduits inside the shell of the heat exchanger which fins enhance heat transfer when utilized in the presence of a two-phase refrigerant. The present invention has application for various heat exchange uses, but is particularly amenable to use in the coil-wound heat exchangers of base load natural gas liquefaction plants.

BACKGROUND OF THE PRIOR ART

Enhanced heat exchange has been a goal sought for many industrial processes. Heat transfer enhancement is particularly attractive in the field of liquefaction of natural gas. Natural gas is a low value fuel resource produced as a by-product from most oil field production operations. The liquefaction of natural gas is necessary for transport from distant production sites to regions having demand for such fuel. Liquefaction is an expensive energy-intensive process. In order to keep the costs of liquefaction for a unit of natural gas low, natural gas recovery and liquefaction are performed only where relatively high production rates of natural gas are available. As a result, base load liquefaction plants tend to be very large and the attendant coil-wound heat exchangers in such plants have become larger, only constrained to the size limitations for transport from the manufacture site to the site of use.

In this type of heat exchange utility as well as other heat exchange utilities, a refrigerant in the liquid phase which comprises either a pure compound or a mixture of compounds is vaporized inside a heat exchanger containment or shell and outside a series of conduits through which the material (natural gas) to be reduced in temperature is passed. The temperature differential between the conduit wall and the refrigerant is usually too small to support nucleate boiling, but refrigerant vapor is

produced at the liquid/vapor interface of the liquid film forming and flowing over the conduits inside the shell of the heat exchanger. This heat transfer process is called convective vaporization.

Various persons skilled in the art have attempted to enhance heat exchange between a fluid passing internally through a conduit and a fluid passing over the external surface of such a conduit. In U.S. Patent 3,217,799 a method for enhancing the condensation of steam on the outside surface of a conduit is disclosed wherein a ribbed configuration as shown in FIG 4 is set forth. The patent is directed to a steam/water system for condensation rather than convective vaporization.

In U.S. Patent 3,384,154, a heat exchange system is set forth wherein condensation occurs on an outer finned surface of a heat exchange conduit as shown in FIG 6. A nucleate boiling surface having a porous structure is formed on the inside surface of the conduit. A discussion of the effects of surface tension on a liquid film is set forth at column 10, line 46-61 and illustrated in FIG 5. The discussion is limited to condensation and not convective vaporization.

In U.S. Patent 3,455,376, a heat exchange surface is disclosed having ribbed surfaces wherein evaporation occurs along the surface with subsequent condensation of the vapor when it contacts a liquid phase existing outwardly from the ribbed surface. This patent is not directly concerned with a conduit having a fluid to be reduced in temperature passing therein.

U.S. Patent 3,587,730 discloses a heat exchanger having porous layers bonded to the walls of the heat exchanger at the surfaces wherein the porous layer comprises conductive particles bonded together to form pores of capillary size on the heat exchange surface. The claims are directed to a plate-fin type heat exchanger structure having a corrugated surface geometry.

In U.S. Patent 3,779,312, a heat exchange conduit is set forth having a specific undulating geometry on the inside surface of the conduit. The system is designed for steam condensation wherein a single-phase fluid is carried inside the tube. The teaching of this patent is distinct from a two-phase system passing over the exterior of a heat exchange conduit.

U.S. Patent 4,118,944 discloses the use of integral internal fins in the heat exchange tube wherein a refrigerant is vaporized inside the tube against

the finned surface thereof. No dimension configuration is set forth for this particular structure.

In U.S. Patent 4,211,276, a fin-tube type heat exchanger is set forth wherein the roughening of the fin surface is desired in order to enhance the drainage of condensate formed on the fins during heat exchange. Again, this patent is directed to condensation and not convective vaporization. Fin dimension or geometry is not set forth beyond the recitation of the surface roughening requirement of the patent.

U.S. Patent 4,216,819 discloses the use of a single layer of randomly distributed metal bodies bonded to a substrate to provide an increased heat exchange surface for condensation. Surface tension characteristics are set forth as an active phenomenon effecting the enhancement of condensation using the recited surfaces.

In U.S. Patent 4,232,728, a structure for enhancing single-phase or condensing heat transfer coefficients on the inside of tubing by bonding a layer of randomly distributed metal bodies to the inner wall is set forth. Criteria are given for the relative height of the bodies in relation to the tube inside diameter and for the void fraction of the bonded layer. This patent does not address the problem of convective vaporization in a two-phase system.

In Reissue Patent 30077, a heat transfer surface for enhancing nucleate boiling is set forth wherein the surface is grooved at a microscopic density and the grooves are subsequently deformed to form restricted openings therein. The restricted openings are the key to the effectiveness of the heat transfer surface. The teaching of this geometry for use in nucleate boiling is dissimilar from the present invention's interest in convective vaporization.

Various literature articles have been directed to an understanding of the physical dynamic parameters of liquid on a fluted or finned surface. Exemplary of such a disclosure is the article, Analysis of Nusselt-Type Condensation on a Vertical Fluted Surface, C. B. Panchal and K. J. Bell, NUMERICAL HEAT TRANSFER, volume 3, pages 357-371, 1980. Such studies again are directed to condensation and not convective vaporization.

The present invention overcomes the limitation of the prior-art heat exchange technology by providing for an ultra-low fin geometry which is controlled by various geometric dimensions, as well as physical properties of

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the refrigerant being utilized in the heat exchange. The utilization of these dimensions and properties in a unique relationship provide for a fin geometry having unexpectedly high heat transfer enhancement. The use of such heat transfer enhancement allows for the reduction in size of heat exchangers or the maintenance of heat exchanger size with increased heat exchange capacity. Such a result is beneficial to heat exchange in general, but is particularly beneficial to the specific application of base load natural gas liquefaction wherein heat exchangers are presently at a near maximum in size and yet economics would dictate that even larger heat exchangers may be desirable. The present invention would provide the increased heat exchange capacity for a relatively smaller heat exchanger size.

BRIEF SUMMARY OF THE INVENTION

The present invention is directed to a heat exchanger having at least one tubular conduit for conducting a fluid from which heat is to be removed through such a heat exchanger wherein a shell surrounding such conduit defines a refrigerant space between said shell and said conduit. The conduit is aligned such that a two-phase refrigerant can pass over the conduit. The conduit includes ultra-low fins affixed outward from the outer surface of said conduit wherein the fin height (H), the fin crest width (w) and the fin gap width (W) are selected in relationship to the density and surface tension of the refrigerant such that;

$$1 \leq \frac{2\,\sigma\,g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) \leq 200$$

The present invention is particularly directed to a heat exchanger having a plurality of coil-wound tubular conduits such that a two-phase refrigerant can pass substantially perpendicular to the axis of said conduits wherein the ultra-low fins are affixed radially transverse or helically outward from the outer surface of the conduits. Such a heat exchanger is particularly appropriate for the refrigeration and liquefaction of natural gas as the gas passes through the interior of the finned conduits.

The invention is also concerned with a process for heat exchanging a fluid in a heat exchanger which has at least one tubular conduit for such a fluid and a shell surrounding said conduit which defines a space for a two-phase refrigerant. The refrigerant passes over the outside surface of

the conduit to remove heat from the fluid passing through the conduit. An enhanced level of heat transfer is achieved during the process of heat exchanging by passing the refrigerant over ultra-low fins affixed to the conduit outer surface wherein the fin height (H), the fin crest width (w) and the fin gap width (W) are selected in relationship to the density and surface tension of the refrigerant such that;

$$1 \leq \frac{2\,\sigma\,g_c}{g(\rho_L - \rho_V)H}\left(\frac{1}{w} + \frac{1}{W}\right) \leq 200$$

The process is more specifically directed to a method wherein the refrigerant passes over the outside surface of said conduits in a direction generally perpendicular to the axis of said conduits and in which the refrigerant passes over ultra-low fins which are radially transverse or helically affixed to the outer surface of said conduits.

The invention is more specifically directed to a process for the liquefaction of natural gas in a method as described above.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG 1 is a drawing of the cross section of a fin surface illustrating the fin parameters of the present invention.

FIG 2 is a graph of constant-heat-flux enhancement factors for radial and longitudinal fins on the heat exchange surface.

FIG 3 is a graph of constant-temperature-difference enhancement factors for rolled fin surface.

FIG 4 is a graph of two-phase pressure drop for rolled finned tubing and bare tube tubing.

FIG 5A is a schematic illustration of the experimental test equipment used in the invention.

FIG 5B is a perspective view of the cell of the test equipment of the invention.

FIG 6 comprises two cross-sectional views of a photomicrograph of experimental sample F12. Photomicrograph (a) is at 26X magnification and photomicrograph (b) is at 120X magnification.

FIG 7 comprises two photomicrograph cross-sectional views of experimental sample F17. Photomicrograph (a) is at 40X magnification and photomicrograph (b) is at 120X magnification.

FIG 8 comprises two photomicrograph cross-sectional views of experimental sample R2. Photomicrograph (a) is at 20X magnification and photomicrograph (b) is at 60X magnification.

DETAILED DESCRIPTION OF THE INVENTION

The invention is an improved means to chill or condense fluids flowing inside the tubes of a heat exchanger, the improvement comprising ultra-low fins affixed more-or-less radially outwards from the outer surface of the tubes. The heat removed from said fluids is transferred to a liquid refrigerant flowing across and vaporizing on the ultra-low-finned tubes. The extent of enhancement to the shellside heat transfer is remarkable and unexpected, exceeding even the maximum enhancement expected based on the increase in shellside heat transfer area. Because the fins are very small, the improved heat transfer is attained with little or no increase in the shellside pressure drop per unit length along the axis of the shell.

The ultra-low-fin tubing is distinguished from tubing which has been suggested in the prior art as being effective for vaporizing liquids by its very low fin height and relatively high fin density (number of fins per unit length). The relatively small dimensions of the fins cause surface tension forces to become very large in relation to viscous and gravitational forces within the liquid films on the wetted finned tubing. The surface tension forces ensure that extremely thin liquid films are maintained on the sides of the fins, especially near the fin crests, resulting in very high local heat transfer coefficients in this region.

The most effective fins are disposed approximately radially transverse on the tubing and can be integrally formed from the tubing base metal or be attached by other methods such as soldering, welding or tension winding. Fins tested and found effective to various extents had trapezoidal, rectangular, or essentially triangular cross sections. Other related fin shapes are expected to be similarly effective. The fins may have flats at either the fin crests or between the fins at the fin root diameter, said flats being more or less parallel with the axis of the tubing. The fin crests and valley bottoms between fins might also be rounded rather than flat. The fins can be made using any of a number of techniques which have been developed and which are available in the art for making conventional finned tubing. Usually, for convenience, such fins are helically disposed

around the tube and can consist of one or more separate helical elements. The fin dimensions found to be important in the practice of the present invention are the fin height (H), the effective width of the fin crests (w) and the width of the gap between adjacent fin crests (W). As will be shown below, practice of the present invention requires that the physical properties of the refrigerant be taken into account in assigning numerical values to these fin parameters.

For approximately radially transverse disposed fins, a remarkable and unexpected enhancement to the shellside heat transfer is found for downflow-vaporization (two-phase flow) conditions. The enhancement can be several times larger than that which might be expected on the basis of the increase in surface area alone. Under the same conditions, longitudinally disposed fins were found to be less effective than radially transverse disposed fins, but were nevertheless still considerably more effective than the prior-art bare tubing. To ensure uniform shellside two-phase flow distribution, the axis of a coil-wound heat exchanger must be vertical. The tubes are helically wound in layers from one end of the exchanger to the other. The angle of winding is small, such that the tubes can be considered to be essentially horizontal. The poorer performance of longitudinally finned tubes is probably caused by their poor characteristics for feed and drainage of liquid films when the tubes are at or close to horizontal. The longitudinal fins might tend to cause flooding in the valleys between fins on the upper part of the tubes, leading to less-effective heat transfer there, and, at the same time, hinder the feed of liquid to the lower part of the tubes, leading to poorer heat transfer there as well. Radially transverse or helically disposed fins facilitate drainage of excess liquid around the tube, leading to much better heat transfer performance. Helically finned tubes are probably best for actual use. Helical fins which approach close to radially transverse fins are preferred.

Two-phase pressure drop measurements on bundles of the proposed ultra-low-fin tubing indicate that the large enhancement to heat transfer can be realized with little or no penalty in pressure drop per unit length of bundle. This is because the fins are small and the major component of pressure drop for flow across tube bundles is due to form drag, not friction drag. Form drag is resistance to flow around a blunt object and is present whether the tubing is finned or bare. Moreover, the enhanced heat transfer

permits shorter exchanger bundles for a given heat duty, with a consequent reduction in the overall shellside pressure drop across the bundle. This can result in significant savings in both the capital and operating costs of the refrigerant compressors.

Description of Geometric Parameters and Operative Limits of Ultra-Low-Fin Tubing

Figure 1 shows the outer finned portion of a longitudinal cross section which contains the axis of a tube which has helically disposed ultra-low fins of trapezoidal shape. The surface illustrated schematically in Figure 1 is that of an actual tube sample (designated below as sample R2) prepared and tested by the inventors. It was produced by a three-wheel roller-head die which formed three continuous side-by-side helical fins with a small winding angle about the tube of 3.5°. Also illustrated in Figure 1 are the geometric parameters of the fins (which apply as well for fins of rectangular, triangular and other related shapes) and the approximate shape or profile of the refrigerant liquid/vapor meniscus on the fins. Of course, the actual profile of the meniscus is constantly fluctuating because of the time-varying nature of two-phase flow and the effects of impinging liquid droplets falling from the tube above and droplets deposited on the tube or sheared off the tube by the vapor phase of the refrigerant, which moves through the heat exchanger at a higher velocity than the liquid phase. However, for purposes of the analysis below, the meniscus profile shown in Figure 1 will serve as a reasonable average.

The nomenclature below defines the geometric parameters of the fins and other parameters of importance in the practice of this invention:

b = width of fin at the fin root diameter

B = width of flat or gap at the fin root diameter

D_o = outer diameter of tube

FD = fin density or number of fins per unit length measured at the tube outer diameter D_o along a direction normal to the direction of the fins. For radially transverse disposed fins and for helically disposed fins with a small angle of winding about the tube, FD is measured in a direction parallel with the axis of the tube; for longitudinally disposed fins, FD is measured in the tube circumferential direction.

g = normal acceleration of gravity, 32.17 ft/sec^2

g_c = Newton's Law conversion factor, 32.17 ft-lb_m/lb_f-sec^2

H = height of fin

w = effective width of fin crest at the tube outer diameter D_o

W = width of gap between adjacent fin crests at the tube outer diameter D_o

α = base angle of fin cross section; for rectangular fins, α = 90°

β = helical fin inclination about tube. For <u>true</u> radially transverse fins, β = 0°; for longitudinal fins, β = 90°.

ρ_L = density of refrigerant liquid phase, e.g. lb_m/ft^3

ρ_V = density of refrigerant vapor phase, e.g. lb_m/ft^3

σ = surface tension of refrigerant liquid, e.g. lb_f/ft or dynes/cm

The operative limits which are pertinent are those associated with the fins themselves. The fin dimensions H, W and w defined above are important.

If the fin height H is too high, the following problems are encountered, although not necessarily at the same fin height:

- The heat conduction path along the fin is lengthened and fin efficiency decreases to the extent that the effectiveness of the finned tubing is reduced.

- Pressure drop across the finned tube increases because of the increased contribution of frictional drag compared to form drag.

- The surface tension mechanism, which is the cause of the unexpectedly large heat transfer enhancement under two-phase downflow-vaporization conditions, would not be effective if the fin height was too large. In this case, the film-thinning action of the surface tension forces would be relegated to an insignificantly small region near the fin crest, rather than acting more-or-less uniformly over much of the height of the fin.

Making the fin height too small also presents problems:

- Under two-phase conditions, the spaces between the fins would flood completely with liquid and the effectiveness of the finned surface would be impaired.

- For single-phase conditions, as the fin height is decreased while maintaining a constant tube outside diameter and a constant flow rate, the fins will eventually become comparable in size to the thickness of the thermal boundary layer on the tubing. As fin height is reduced further, the boundary layers completely swamp the fins, and the thermal performance of the finned tubing approaches that of bare tubing, yielding no advantage to heat transfer.

Similarly, if the gap W between the fin crests is too large, the following difficulties ensue:

- As W is increased, the magnitude is decreased of the surface tension forces which pull excess refrigerant liquid from the sides of the fins into the valleys, leading to less effective heat transfer.

- Also, increasing W is equivalent to moving the fins farther apart, which decreases the heat transfer area per unit length of tubing.

If W is made too small, too much of the space between the fins will be filled with slow-draining liquid, again reducing the effectiveness of the heat transfer process.

If the width of the fin crest w is too large, the following adverse effects occur:

- As w is increased, the magnitude is decreased of the surface tension forces which push excess refrigerant liquid from the crests onto the sides of the fins, leading to less effective heat transfer.

- Also, increasing w decreases the surface area per unit length of tubing.

If the crest width w is made too small, the film-thinning effect in the immediate vicinity of the crest region will be large, but the narrow tips of the fins will then present too much resistance to conduction of heat to the fin tips, leading to decreasing heat transfer effectiveness as w is decreased further. It should be noted that, whereas triangular-shaped fins in principle have a fin crest w equal to zero, in practice there is always a finite crest width due to the limitations of fin-forming operations (machining, roll forming, extruding, etc.).

It is seen from the above that, for successful practice of this invention, lower and upper bounds must be placed on H, W and w. As will be shown below, these bounds must also take into account the physical properties σ, ρ_L and ρ_V of the refrigerant.

It is well known that fin efficiency decreases with an increase in fin height or heat transfer coefficient and increases with an increase in fin thermal conductivity. The fins should be made from a metal with high thermal conductivity so that fin efficiencies remain high even for the high heat transfer coefficients obtained with the present invention. Preferred metals for the fins are copper, brass, aluminum or aluminum alloys.

It is assumed that the refrigerant liquid wets the finned tube well (contact angle between the liquid and metal is small) and has a viscosity low enough so that viscous forces within the liquid film can be considered negligible compared with surface tension forces. Common refrigerants, the lower-molecular-weight hydrocarbons, cryogenic liquids and many other liquids satisfy these requirements.

Definition of Heat Transfer Enhancement Factors

The heat transfer coefficients ascertained for the finned surfaces of the examples which were tested under two-phase conditions are termed effective coefficients since, in all cases, the coefficients were referenced to the surface area of a bare tube having the same outside diameter (D_o) as the finned surface. The enhancement factor for a finned tube is defined as the ratio of the effective heat transfer coefficient for the finned tubing to the heat transfer coefficient for bare tubing. The finned and bare tubing samples which were tested all had essentially the same outside diameters.

The following definitions are made:

h_f = effective heat transfer coefficient for finned tubing referenced to the outside surface area of a bare tube with the same OD (D_o) as the finned tubing

h_b = heat transfer coefficient for bare tubing referenced to the outside surface area of bare tubing

e_q = heat transfer enhancement factor at constant heat flux = h_f/h_b, both coefficients evaluated at the same heat flux

$e_{\Delta T}$ = heat transfer enhancement factor at constant wall-to-fluid temperature difference = h_f/h_b, both coefficients evaluated at the same temperature difference

By definition, the heat flux q is related to h by

$$q = h\,\Delta T \qquad (1)$$

It follows that e_q and $e_{\Delta T}$ are given by

$$e_q = \left(\frac{h_f}{h_b}\right)_q = \left(\frac{\Delta T_b}{\Delta T_f}\right)_q \qquad (2)$$

$$e_{\Delta T} = \left(\frac{h_f}{h_b}\right)_{\Delta T} = \left(\frac{q_f}{q_b}\right)_{\Delta T} \qquad (3)$$

Also,

A_o = outside surface area per unit length of finned tubing; e.g., in^2/in

$(A_o)_{bare}$ = outside surface area per unit length of bare tubing with the same OD as the finned tubing; e.g., in^2/in

= πD_o

Conventional practice in other heat transfer processes, such as condensation, used fins which were much taller than those of the invention, and it was believed that liquid films would essentially engulf the fins, leading to poor heat transfer performance. At the very best, conventional practice suggested that the heat transfer performance would be characterized by

$$e_{\Delta T} \text{ (or } e_q \text{)} \leq A_o/(A_o)_{bare} \qquad (4)$$

and that the enhancement factor could equal the area ratio $A_o/(A_o)_{bare}$ only if all of the surface area was effective and if the finned-tube heat transfer coefficient based on the actual finned-tube area (A_o) was not somehow reduced in magnitude due to the very presence of the fins, as it is sometimes for single-phase flow.

However, the experimental program of the invention led to the <u>unexpected</u> finding that the fin parameters could be chosen to give

$$e_{\Delta T} \text{ (or } e_q \text{) } > A_o/(A_o)_{bare} \qquad (5)$$

The extent of enhancement measured was remarkable, as well as unexpected.

<u>Criteria for Surface-Tension-Augmented Heat Transfer on Ultra-Low-Fin Tubing</u>

It was mentioned above that the physical properties σ, ρ_L and ρ_V of the refrigerant must be considered in practicing the present invention. It will now be shown how these physical properties are taken into account. Consider the refrigerant liquid/vapor meniscus shown in Figure 1.

The difference in pressure between a liquid and vapor at any point on a curved liquid/vapor interface is given by Laplace's equation:

$$P_L - P_V = - \sigma \left(\frac{1}{R_1} + \frac{1}{R_2}\right) \qquad (6)$$

The principal radii of curvature R_1 and R_2 are defined to be positive within the vapor phase and lie in any two mutually perpendicular planes whose line of intersection defines the normal to the point in question on the interface. If either radius of curvature lies within the liquid its sign becomes negative. In applying Equation (6) to the problem at hand, it is assumed that curvature effects in the direction normal to Figure 1 can be neglected; i.e., in the direction parallel with the fin orientation. Accordingly, one of the two radii can be set equal to infinity, giving

$$P_L - P_V = - \frac{\sigma}{R} \qquad (7)$$

where R is now the local radius of curvature of the liquid/vapor interface in a plane normal to the fin orientation. The vapor pressure P_V is constant and is equal to the bulk pressure of the two-phase stream flowing past the ultra-low-finned tube. Therefore, Equation (7) gives the local variation of the pressure within the liquid film from the vapor-phase or bulk pressure. At the crests of the fins, R is negative (convex liquid film) and Equation (7) indicates that P_L is greater than P_V. At the lowest point on the meniscus in the valleys between the fins, R is positive (concave liquid film) and P_L is less than P_V. The resulting pressure gradient within the liquid film causes very rapid draining of any liquid which impinges on the fin crests to the valleys between the fins.

The magnitude of the pressure gradient within the liquid film can be estimated from the fin parameters. If R_c and R_v are the characteristic radii of curvature of the liquid film at the fin crest and valley, respectively, these radii can be represented by

$$R_c = -\frac{w}{2} \qquad (8)$$

$$R_v = \frac{W}{2} \qquad (9)$$

and Equation (7), applied at the fin crest and valley, gives the following relation for the pressure difference within the liquid film between fin crest and valley:

$$\text{surface-tension-induced pressure difference} = 2\,\sigma \left(\frac{1}{w} + \frac{1}{W}\right) \qquad (10)$$

Because this pressure difference acts over a distance approximately equal to the fin height H, the surface-tension-induced pressure gradient within the liquid film is given by

$$\text{surface-tension-induced pressure gradient} = \frac{2\,\sigma}{H} \left(\frac{1}{w} + \frac{1}{W}\right) \qquad (11)$$

It is shown below that these surface-tension-induced pressure gradients can be very much larger than either gravitational or shear (pressure drop) forces. The resulting very thin films of liquid on the sides of the upper part of the fins results in extremely high heat transfer coefficients since the local heat transfer coefficient is commonly assumed to be inversely proportional to the local thickness of the liquid film. The bulk of the liquid will drain in the valleys between fins, giving lower heat transfer coefficients in the valleys than near the fin crests. However, the net effect can be an enhancement to heat transfer which is much larger than that expected based on the increase in surface area over that of bare tubing; i.e.,

$$e_{\Delta T} \text{ (or } e_q) > A_o/(A_o)_{bare} \qquad (5)$$

Based on the above, it is expected that the most effective fin geometries would be those for which the surface-tension-induced pressure gradients within the liquid films are greater than the net gravitational pressure gradient within the liquid films; i.e.,

$$\frac{\text{surface-tension-induced pressure gradient}}{\text{net gravitational pressure gradient}} > 1 \qquad (12)$$

The denominator of Equation (12) is uniquely determined from the liquid and vapor densities and the local acceleration of gravity and is given by

$$\text{net gravitational pressure gradient} = (\rho_L - \rho_V)\frac{g}{g_c} \qquad (13)$$

Using Equations (11) and (13), the criterion Equation (12) then becomes

$$\frac{2\,\sigma\, g_c}{g\,(\rho_L - \rho_V)\,H}\left(\frac{1}{w} + \frac{1}{W}\right) > 1 \qquad (14)$$

For ultra-low-finned surfaces and refrigerants which satisfy Equation (14), surface tension enhancement should occur at all points on the circumference of the tube because the liquid films can then be thinned even in opposition to the pull of gravity. For example, at the bottom of a horizontal or near-horizontal tube drainage will likely take

place from the fin crests but thin films of liquid can still be pulled against the force of gravity up the sides of the fins to give high heat transfer coefficients.

Table 1 lists the parameters of four finned tubes which have been tested for downflow-vaporization of R-11 refrigerant, composed of a halogenated hydrocarbon, under conditions set forth in the following example.

TABLE 1

Parameters of the Finned Tubes Tested(1)

Sample	Fin Cross Section	FD, fins/in.	D_o, in.	b, in.	B, in.	w, in.	W, in.	H, in.	α degrees	β degrees	A_o, in.^{2}/in.
F12, machined helical fins	trapezoidal(3)	92.6	0.3775	0.0095	0.0014	0.0010	0.0098	0.0228	79.4	0.56	4.65
F17, machined longitudinal fins	trapezoidal(3)	84.2	0.378	0.0073	0.0020	0.0020	0.0085	0.0220	83.1	90	5.43
RF2, tension-wound copper fins on aluminum tube	rectangular	57.1	0.3714	0.0049	0.01262	0.0049	0.01262	0.0291	90	0.93	4.25
R2, rolled fins(2)	trapezoidal	43.5	0.3930	0.0202	0.0031	0.0087	0.0143	0.0264	77.7	3.5	3.02

Notes: (1) All samples were made of aluminum except as noted.

(2) Three continuous helical fins made with a three-wheel roller-head die.

(3) Although trapezoidal in cross section, the fins of samples F12 and F17 were nearly triangular in cross section.

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EXAMPLE

The ultra-low-finned tubes, as well as bare tubes, were tested in in-line, square-pitch tube bundles in an apparatus as illustrated in FIG 5A and B. The vapor quality (weight fraction vapor) of the flowing refrigerant was varied from zero (all liquid) to 0.9. Heat was supplied to the tubes with an electric cartridge heater, and the temperature difference between a known radial position in the wall of the tube and the fluid was measured directly and accurately using an opposed thermopile circuit. The instrumented tubes could be rotated about their axis within the tube bundle. This allowed the wall-to-fluid temperature difference to be averaged at many points around the circumference of the tubing. A correction was then made for the wall temperature drop to obtain an average surface temperature, and the average heat transfer coefficient for the surface was calculated from the known heat flux q and the measured temperature difference between the surface and the fluid.

As shown in FIG 5B, the tube bundle 10 was mounted in a hollow cell 16 such that a series of half- and full-diameter dummy tubes 12 and 14, respectively, surrounded the test tube 18 in order to place the tube in an accurate refrigerant flow environment. Vapor-phase refrigerant entered the top of the cell 16 through an inlet 20 and liquid-phase refrigerant entered the cell 16 through a distributor 22. The two phases mix in a plenum 24 and pass downwardly around the various tubes, exiting through a bottom outlet 28.

The refrigerant is then recycled for flow through the cell 16 in the apparatus shown in FIG 5A. The cell 16 is shown with the vapor and liquid inlets 20 and 22. The discharge two-phase refrigerant from outlet 28 is passed to a separation vessel 30 wherein all of the vapor phase is removed to be re-condensed against cooling water in heat exchanger 32. The water circulates through conduits 34 and 36.

Refrigerant liquid from the bottom of vessel 30 passes to a heated reboiler 38 wherein vapor is regenerated and passes through line 40 and flowmeter 42. Liquid is circulated via line 44 and pump 46 through a second flowmeter 48. In this manner, an accurate experimental environment was provided to test various samples of ultra-low-finned tube for the enhanced heat transfer effect set forth herein.

Figures 6-8 are scanning electron microscope (SEM) photomicrographs of the fins on the tubes tested. Each photograph is oriented so that the metal wall of the ultra-low-fin tubing is at the bottom of the photograph.

Figure 6 shows two photomicrographs of a cross section the plane of which contains the axis of sample F12, said sample having a single helically disposed ultra-low fin, said fin being generally trapezoidal, but nearly triangular, in cross section. Figure 6A shows the fins at 26X magnification and Figure 6B shows the fins at 120X magnification.

Figure 7 shows two photomicrographs of a cross section the plane of which passes normal to the axis of sample F17, said sample having 100 longitudinally disposed ultra-low fins on the circumference of the sample, said fins being generally trapezoidal, but nearly triangular, in cross section. Figure 7A shows the fins at 40X magnification, while Figure 7B shows the fins at 120X magnification.

Figure 8 shows two photomicrographs of a cross section the plane of which contains the axis of sample R2, said sample having three contiguous helically disposed ultra-low fins, said fins being generally trapezoidal in cross section. Figure 8A shows the fins at 20X magnification. Figure 8B shows the fins at 60X magnification.

Figure 2 shows the measured constant-heat-flux enhancement factor e_q for finned surfaces F12 and F17 which constitute examples of the invention. Both of these surfaces had fins of approximately similar shape and dimensions. Although the longitudinally finned sample F17 had

nearly 17% more surface area than the approximately radially finned sample F12, sample F12 gave clearly superior performance and illustrates the unexpected degree of enhancement which is the main focus of this invention; i.e., sample F12 has

$$e_q > A_o/(A_o)_{bare} \quad (15)$$

On the other hand, sample F17 with longitudinal fins has

$$e_q < A_o/(A_o)_{bare} \quad (16)$$

Figure 2 also shows that sample F12's unexpected degree of enhancement continues to persist even at vapor qualities as large as 0.9, indicating that surface tension forces are probably very large compared to vapor-shear effects. This is demonstrated below for sample R2.

Figure 3 shows the measured constant-temperature-difference enhancement factor $e_{\Delta T}$ for the rolled-fin sample R2. Over a wide range of conditions, this surface had enhancement factors which were unexpectedly greater than $A_o/(A_o)_{bare}$. This area ratio was 2.45 for sample R2. Values of $e_{\Delta T}$ as large as 7.1 were measured for $\Delta T = 0.3°F$. This is almost 200% larger than the <u>maximum</u> enhancement which conventional practice would lead one to believe is possible; i.e.,

$$(e_{\Delta T} \text{ or } e_q)_{maximum} = A_o/(A_o)_{bare} \quad (17)$$

It is significant to note from Figure 3 that, for vapor qualities less than 0.9, the enhancement factor increases as the surface-to-bulk temperature difference decreases. This makes the invention particularly attractive for cryogenic heat transfer applications, which operate generally at small temperature differences.

The surface-tension-induced pressure gradient within the liquid film is calculated below for sample R2 and compared with net gravitational and two-phase-flow pressure gradients. Restating Equation (11), we have

$$\text{surface-tension-induced pressure gradient} = \frac{2\,\sigma}{H}\left(\frac{1}{w} + \frac{1}{W}\right) \tag{11}$$

where, for sample R2 and refrigerant R-11 at 76°F, we have

$$\text{surface-tension-induced pressure gradient} =$$

$$\frac{(2)\ (18.34\ \text{dynes/cm})}{(0.0264\ \text{in.})\ (2.54\ \text{cm/in.})}\left(\frac{1}{0.0087\ \text{in.}} + \frac{1}{0.0143\ \text{in.}}\right) \times 10^{-6}\ \frac{\text{bar}}{\text{dynes/cm}^2} \times 14.5\ \frac{\text{psi}}{\text{bar}}$$

$$= 1.47\ \text{psi/inch}$$

This is a very respectable pressure gradient and, in fact, is much larger than pressure gradients normally used in the design of liquid-flow conduits. The net gravitational pressure gradient associated with a column of R-11 liquid can be shown to be small compared with this surface-tension-induced pressure gradient. At 76°F, this net gravitational gradient is found from Equation (13) to be

$$(\rho_L - \rho_V)\,\frac{g}{g_c} = \frac{(92.227 - 0.374)(32.17)}{(32.17)(144)(12)} = 0.0532\ \text{psi/inch}$$

This is about 28 times smaller than the surface-tension-induced pressure gradient.

It can be similarly shown that the pressure gradient for two-phase flow through a bundle of the rolled-fin tubing is also small compared with the surface-tension-induced pressure gradient. Two-phase, downflow, pressure drop data for a bundle of finned tubes identical with sample R2 are shown in Figure 4. The largest pressure drop measured for $G = 40000\ lb_m/hr\text{-}ft^2$ was 0.213 inches of water per row of tubes. Given a bundle longitudinal pitch ratio of 1.189 and a tube outside

diameter of 0.3930", the following representative pressure gradient for the rolled-fin bundle can be calculated:

$$\text{Two-phase pressure gradient} = \frac{0.213}{(1.189)(0.3930)(27.76)} = 0.0164 \text{ psi/inch of bundle}$$

Remarkably, this is about 90 times smaller than the surface-tension-induced pressure gradient. If it is recognized that most of the two-phase pressure drop is probably associated with form drag, drag due to flow around a blunt object and present whether the tubes are finned or bare, the actual pressure gradient attributable to vapor shear forces could be smaller than the surface-tension-induced pressure gradient by a factor much larger than 90.

Figure 4 is a graph showing pressure drop data for downward two-phase flow of refrigerant-11 over a bare tube bundle and a bundle of rolled-fin tubing identical to ultra-low-fin sample R2. These data show that the very significant heat transfer enhancement of the present invention can be obtained with no significant penalty in pressure drop per unit length of bundle. Moreover, for a given heat transfer duty, the enhanced heat transfer will allow shorter bundle lengths to be used, with a consequent significant <u>reduction</u> in the overall pressure drop across the bundle.

Table 2 summarizes various calculated parameters for the four finned tubes of the examples which were tested, as well as the measured heat transfer enhancement factors for these surfaces. The helically finned samples F12, RF2 and R2 all exhibited unexpectedly large enhancement factors. Practically, the fins on these three samples can be considered to be close to true radial transverse fins because of their small helical winding angle β. For sample RF2, the extent to which the enhancement factor exceeded $A_o/(A_o)_{bare}$ was small, but nevertheless significant. The copper fins on this sample were held to the aluminum tube only by tension, so some contact resistance was undoubtedly present. It is likely that the performance of sample RF2 could have been markedly improved by eliminating the contact resistance

by soldering or welding the fins to the tube. Sample F17 with longitudinal fins was the only one to exhibit enhancement factors which were less than $A_o/(A_o)_{bare}$. A possible explanation for the poorer performance of longitudinally finned tubes was given earlier.

TABLE 2

Calculated and Experimental Parameters of the Finned Tubes Tested for Downflow Vaporization of R-11 Refrigerant (1)

Sample	FD Fins/Inch	$\frac{2\sigma g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right)$	$A_o/(A_o)_{bare}$	$e_{\Delta T}$	e_q
F12, machined helical fins	92.6	190	3.92(2)	—	4.4 to 5.1
F17, machined longitudinal fins	84.2	111	4.57	—	1.75 to 2.6
RF2, tension-wound copper fins on aluminum tube	57.1	38.4	3.64	2.5 to 4.3(5)	—
R2, rolled fins	43.5	27.6	2.45	2.0 to 7.1(3)	3 to 6(4)

Notes:

(1) Properties of R-11 refrigerant evaluated at 76°F.

(2) $(A_o)_{bare} = \pi D_o$ surface area per unit length of a bare tube with the same outside diameter as the finned tube. This convention was used for all samples.

(3) For ΔT between 0.3 and 13°F.

(4) For q between 1000 and 5000 BTU/hr-ft^2.

(5) For ΔT between 0.3 and 10°F.

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It can be seen from Table 2 that all of the approximately radially finned surfaces which exhibited unexpectedly large enhancement factors also satisfied Equation (14); i.e.,

$$\frac{2\,\sigma\,g_c}{g\,(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) > 1 \qquad (14)$$

Equation (14) can be considered a necessary condition for the successful practice of the present invention; however, it is not a sufficient condition. For example, ultra-low fins designed for the vaporization of a given refrigerant according to Equation (14) will not display the high enhancements of the present invention if the ultra-low-finned tubing is totally immersed in the liquid refrigerant. The liquid phase of the refrigerant must be deposited on the tubing in the form of thin films of liquid. Also, while it is clear that, for a given refrigerant and fin shape, the left-hand side of Equation (14) can be made as large as desired by decreasing the fin dimensions (H, w and W), an upper limit must be placed on the left-hand side of Equation (14) to avoid completely flooding the valleys between the fins with the liquid refrigerant. The experiments, summarized in Table 2, and the remarks presented herein lead to the following criterion to obtain the unusual extent of enhancement of the present invention:

$$1 \leq \frac{2\,\sigma\,g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) \leq 200 \qquad (18)$$

In using this criterion, the refrigerant properties σ, ρ_L and ρ_V determine the size or height of a fin of given geometry. Once the refrigerant is chosen, the lower limit in Equation (18) determines the largest fin size for which surface-tension-augmented heat transfer will occur. Similarly, the upper limit in Equation (18) determines the smallest fin size for which the surface tension enhancement mechanism will not be excessively inhibited by liquid flooding between the fins.

Within reasonable variation, the exact shape of the ultra-low fins protruding from the surface of the tubing is not important in the practice of the present invention. However, a trapezoidal fin cross

section is preferred for coil-wound exchangers because trapezoidal fins are relatively easy to form and can be made sufficiently robust to resist significant deformation during the winding process. Localized deformation of fins at points of contact with other finned tubes or spacer elements is significant only to the extent that dimensional tolerances (clearance between tube layers, ultimate diameter of the coil-wound bundle, etc.) are affected.

For a given refrigerant, it is clear from Equation (18) that there is no single unique fin design which gives the unexpected enhancement of the present invention. Many successful fin configurations are possible depending on the shape of the fin cross section and on the choice of the fin dimensions H, w and W. A good fin design will provide for adequate drainage space between the fins while at the same time maintaining a large surface area per unit length of tubing relative to bare tubing. In the preferred embodiment of the invention, it is recommended that the fin dimensions be chosen such that:

$$w < W < H \qquad (19)$$

The present invention has been set forth with regard to several preferred embodiments. However the scope of the invention should not be ascertained by these embodiments, but rather should be ascertained from the claims which follow.

CLAIMS

1. A heat exchanger having at least one tubular conduit for conducting a fluid from which heat is to be removed through said exchanger, a shell surrounding said conduit defining a refrigerant space between said shell and said conduit, said conduit aligned such that a two-phase refrigerant can pass over said conduit, the improvement comprising ultra-low fins affixed outward from the outer surface of said conduit wherein the fin height (H), the fin crest width (w) and the fin gap width (W) are selected in relationship to the density and surface tension of the refrigerant such that;

$$1 \leq \frac{2\,\sigma\,g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) \leq 200$$

2. A heat exchanger having a plurality of coil-wound tubular conduits for conducting a fluid from which heat is to be removed through said exchanger, a shell surrounding said conduits defining a refrigerant space between said shell and said conduits, said conduits aligned such that a two-phase refrigerant can pass substantially perpendicularly to the axis of said conduits, the improvement comprising ultra-low fins affixed radially transverse or helically outward from the outer surface of said conduits wherein the fin height (H), the fin crest width (w) and the fin gap width (W) are selected in relationship to the density and surface tension of the refrigerant such that;

$$1 \leq \frac{2\,\sigma\,g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) \leq 200$$

3. The apparatus of Claim 2 wherein the fluid from which heat is to be removed is natural gas.

4. The apparatus of Claim 2 wherein $w < W < H$.

5. A process for removing heat from a fluid in a heat exchanger having at least one tubular conduit for conducting said fluid through said exchanger and a shell surrounding said conduit defining a space for a two-phase refrigerant wherein the refrigerant passes over the outside surface of said

conduit to remove heat from said fluid flowing in said conduit, the improvement comprising an enhanced level of heat transfer achieved by passing the refrigerant over a conduit which has ultra-low fins affixed thereto in which the fin height (H), the fin crest width (w) and fin gap width (W) are selected in relationship to the density and surface tension of the refrigerant such that;

$$1 \leq \frac{2\,\sigma\, g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) \leq 200$$

6. A process for removing heat from a fluid in a heat exchanger having a plurality of coil-wound tubular conduits for conducting said fluid through said exchanger and a shell surrounding said conduits defining a space for a two-phase refrigerant wherein the refrigerant passes over the outside surface of said conduits in a direction generally perpendicular to the axis of said conduits such as to remove heat from said fluid flowing in said conduits, the improvement comprising an enhanced level of heat transfer achieved by passing the refrigerant over conduits which have ultra-low fins radially transverse or helically affixed thereto in which the fin height (H), the fin crest width (w) and fin gap width (W) are selected in relationship to the density and surface tension of the refrigerant such that;

$$1 \leq \frac{2\,\sigma\, g_c}{g(\rho_L-\rho_V)H}\left(\frac{1}{w}+\frac{1}{W}\right) \leq 200$$

7. The process of Claim 6 wherein the fluid from which heat is to be removed is natural gas.

8. The process of Claim 6 wherein $w < W < H$.

9. The process of Claim 6 wherein the temperature difference between the conduit wall and the bulk refrigerant is less than 13°F.

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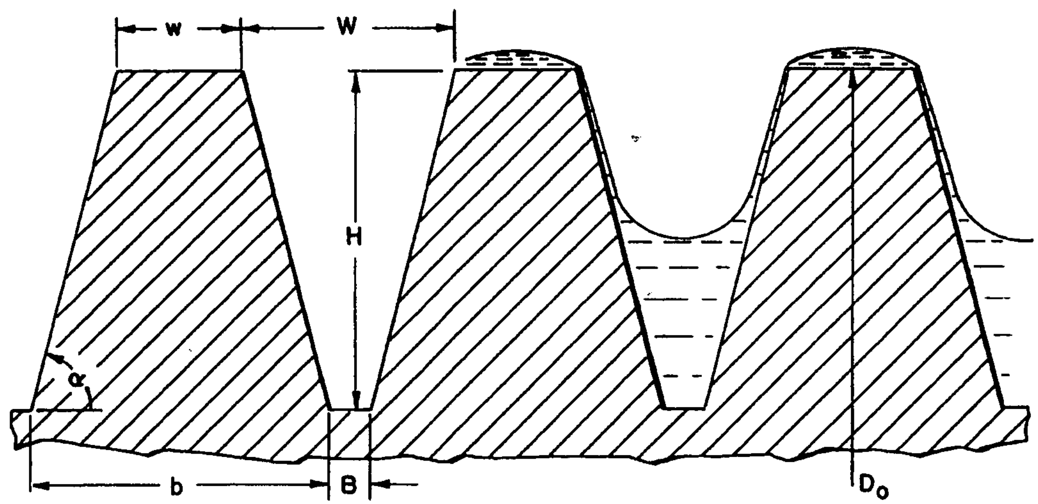
FIG. 1

SCHEMATIC ILLUSTRATION OF ULTRA-LOW FINS
AND DEFINITION OF FIN PARAMETERS



$$FD = \frac{1}{w + W}$$

FIG. 2

CONSTANT-HEAT-FLUX ENHANCEMENT FACTORS FOR RADIAL (F12) AND LONGITUDINAL (F17) FINS

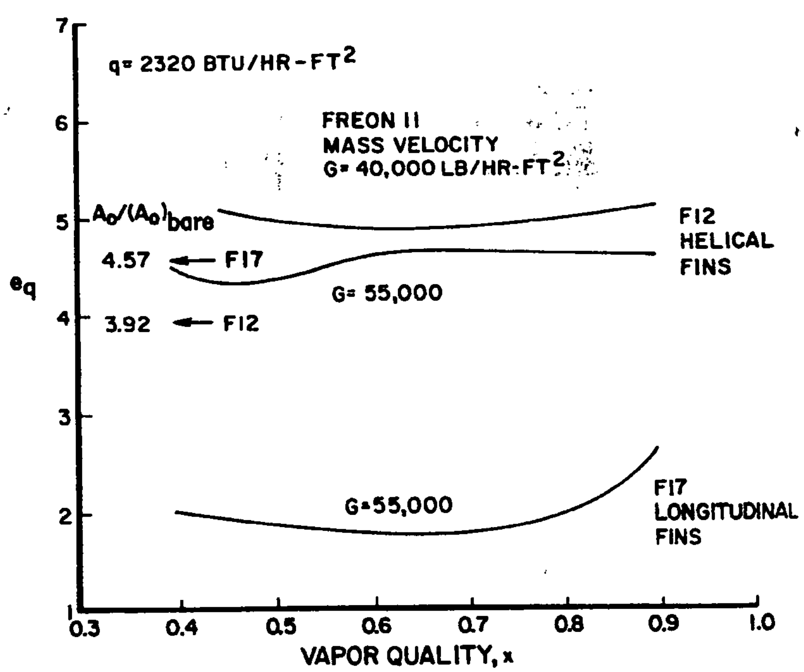


FIG. 3

CONSTANT-TEMPERATURE-DIFFERENCE ENHANCEMENT FACTORS FOR THE ROLLED-FIN SURFACE R2

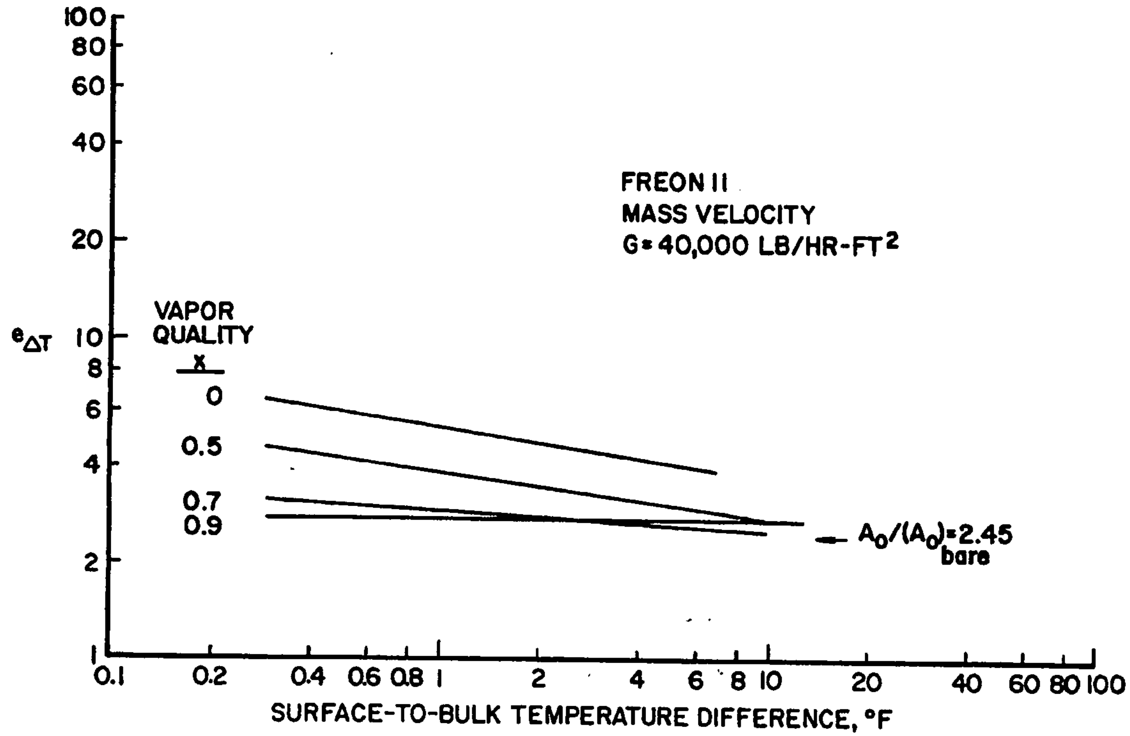
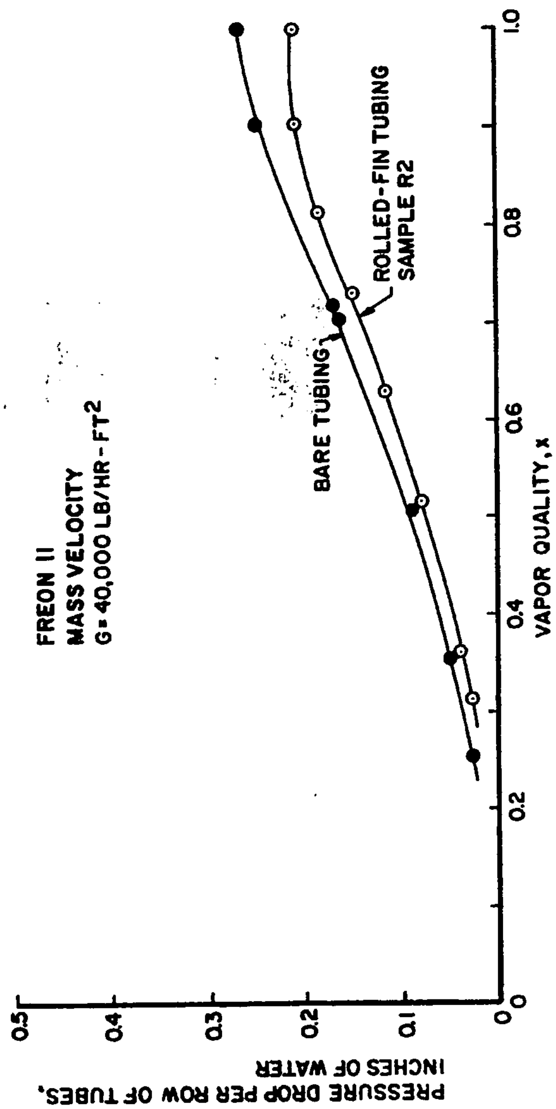


FIG. 4

COMPARISON OF TWO-PHASE PRESSURE DROP IN SQUARE-PITCH BUNDLES OF ROLLED-FIN AND BARE TUBING



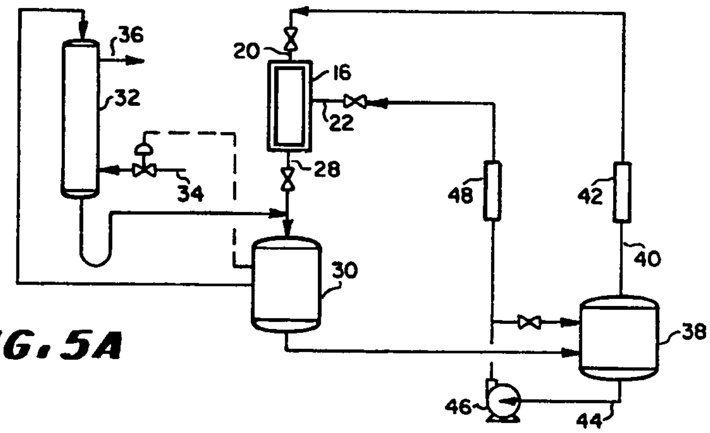
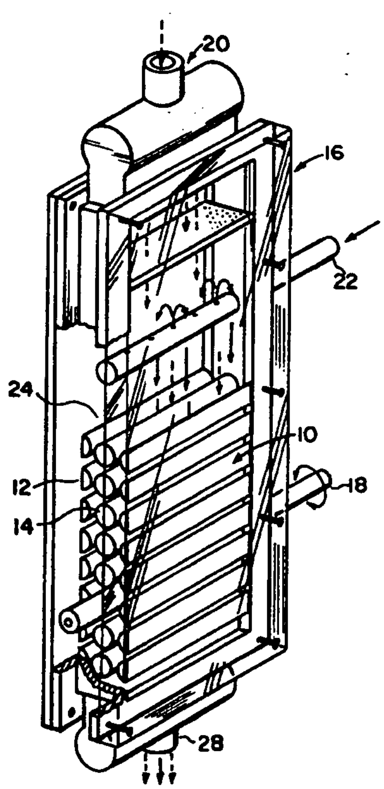


FIG. 5A

FIG. 5B



SEM PHOTOMICROGRAPHS OF SAMPLE F12

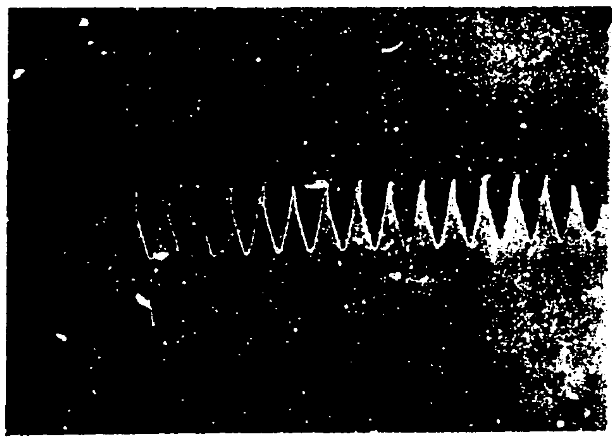


FIG. 6 (a). 26X MAGNIFICATION

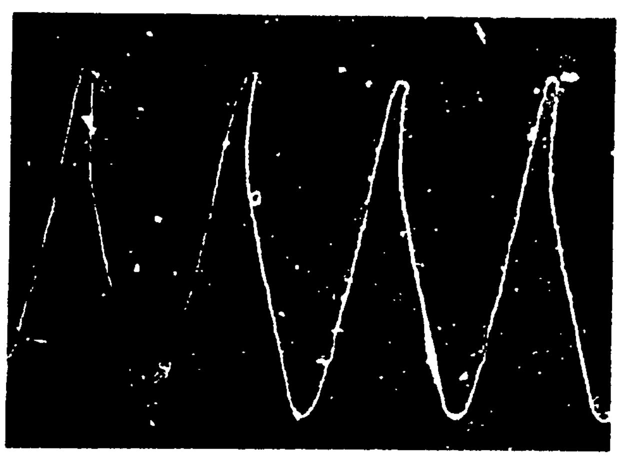


FIG. 6 (b). 120X MAGNIFICATION

SEM PHOTOMICROGRAPHS OF SAMPLE F17

FIG. 7 (a). 40X MAGNIFICATION

FIG. 7 (b). 120X MAGNIFICATION

SEM PHOTOMICROGRAPHS OF SAMPLE R2

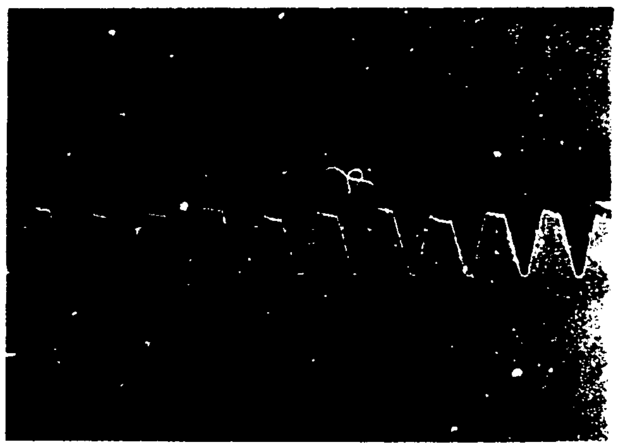


FIG.8 (a). 20X MAGNIFICATION

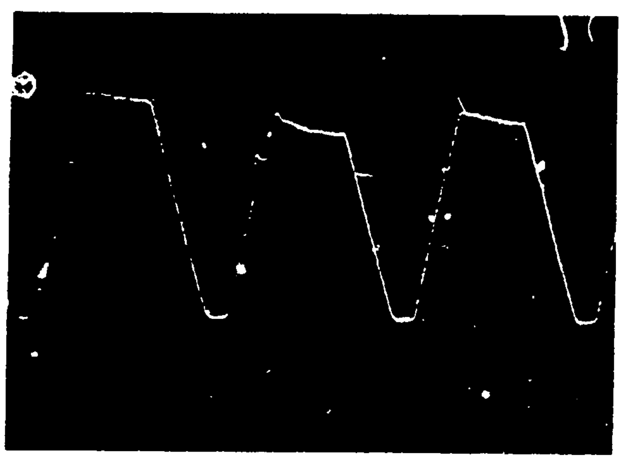


FIG.8 (b). 60X MAGNIFICATION